THE STUDY OF A SHELTER COOLING SYSTEM USING METHANOL AS THE HEAT SINK AND AS THE FUEL

Office of Civil Defense
Work Unit 1422A

BATTELLE MEMORIAL INSTITUTE
Columbus Laboratories
505 King Avenue
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August, 1¢

Dear Sir:

Enclosed is Battelle's research report, "The Study of a Shelter Cooling System Using Methanol as the Heat Sink and as the Fuel", prepared under Subcontrac* No. B-64207(4949A-9)-US, Work Unit 1422A. This report is being sent to you in accordar with the instructions for distribution received from the Office of Civil Defense.

Sincerely,

James A. Eibling

James A. Eibling, Chief
Thermal Systems Division

JAE:pa
Enc.

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THE STUDY OF A SHELTER COOLING SYSTEM USING METHANOL AS THE HEAT SINK AND AS THE FUEL

by

John D. Hummell and W. Donald Beck

INTRODUCTION

A study of both novel and conventional cooling systems was described in a Battelle report to the Office of Civil Defense, "Methods for Disposing of Excess Shelter Heat", by J. D. Hummell, D. E. Bearint, and L. J. Flanigan, under Contract No. OCD-OS-62-191, Subtask 1422A. That report included a section on the open-cycle vapor-compression system using methanol as the working fluid, the heat sink, and the fuel. Possible applications and merits of the system were discussed and compared with alternative cooling methods which could be used under similar shelter cooling circumstances.

This report covers the first phase of additional work under Contract No. B-64207 (4949A-9)-US to evaluate experimentally the potential of the methanol cooling system.

OBJECTIVES

The objectives of the research were to determine:

(1) The adaptability of commercially available compressors as methanol-vapor compressors

(2) The adequacy of commercial lubricants for compressors requiring lubrication.

(3) The evaporator temperature and pressure which would result in the best system performance.

SUMMARY

The data obtained during the operation of a cooling system using a lubricated rotary-vane compressor clearly demonstrated that:

(1) Commercial lubricants are satisfactory for lubricating methanol-vapor compressors.

(2) Compressor performance on methanol vapor is nearly the same as when pumping air.

(3) For the lowest system cost the evaporator pressure was about 1.5 psia, saturation temperature 62 F.

(4) The refrigerating capacity of a commercial finned-tube evaporator when using methanol was about one-half the rated design capacity for typical air-conditioning service.

(5) System costs are comparable to those of alternative systems which could function under the same circumstances.

(6) Further studies should be made of a methanol system using the methanol vapors in an engine as the power source.

THE COOLING SYSTEM

The open-cycle methanol cooling system consists of five major components; me anol supply sufficient to absorb the cooling load by evaporation, a throttling device, direct-expansion evaporator, vapor compressor, blower, and a power source. The l: uid methanol from a storage facility would be throttled into an evaporator at a pressu below 2 psia where the methanol would evaporate to satisfy the cooling load. An engi or electric-motor driven, vapor compressor would maintain the evaporator suction pressure and pump the vapors to a pressure high enough for discharge into the atmosphere, perhaps through an atmospheric-pressure burner to prevent the spread of the toxic vapors.

Figure 1 is a sketch and Figure 2 a photograph of the laboratory cooling system and the test loop used for its evaluation. To provide a test facility which would be insensitive to ambient air temperature and humidity, a closed loop was used for the air flow portion of the laboratory set up. The methanol, evaporator, throttling device, a compressor were typical components which could be used in a shelter cooling system. The remainder of the laboratory system was designed only to provide the needs of the research program.

Methanol

The methanol used for the laboratory studies was a commercial grade obtained i 55-gallon drums from a local distributor of chemicals. During the laboratory work, temperature and pressure measurements verified that the purchased supply had vapor pressure-temperature relations characteristic of methanol.

Throttling Device

A capillary tube on the inlet of each of the 20 coil circuits was used as the throttling device, and also as a metering device to assure uniform distribution of methanol throughout the evaporator.

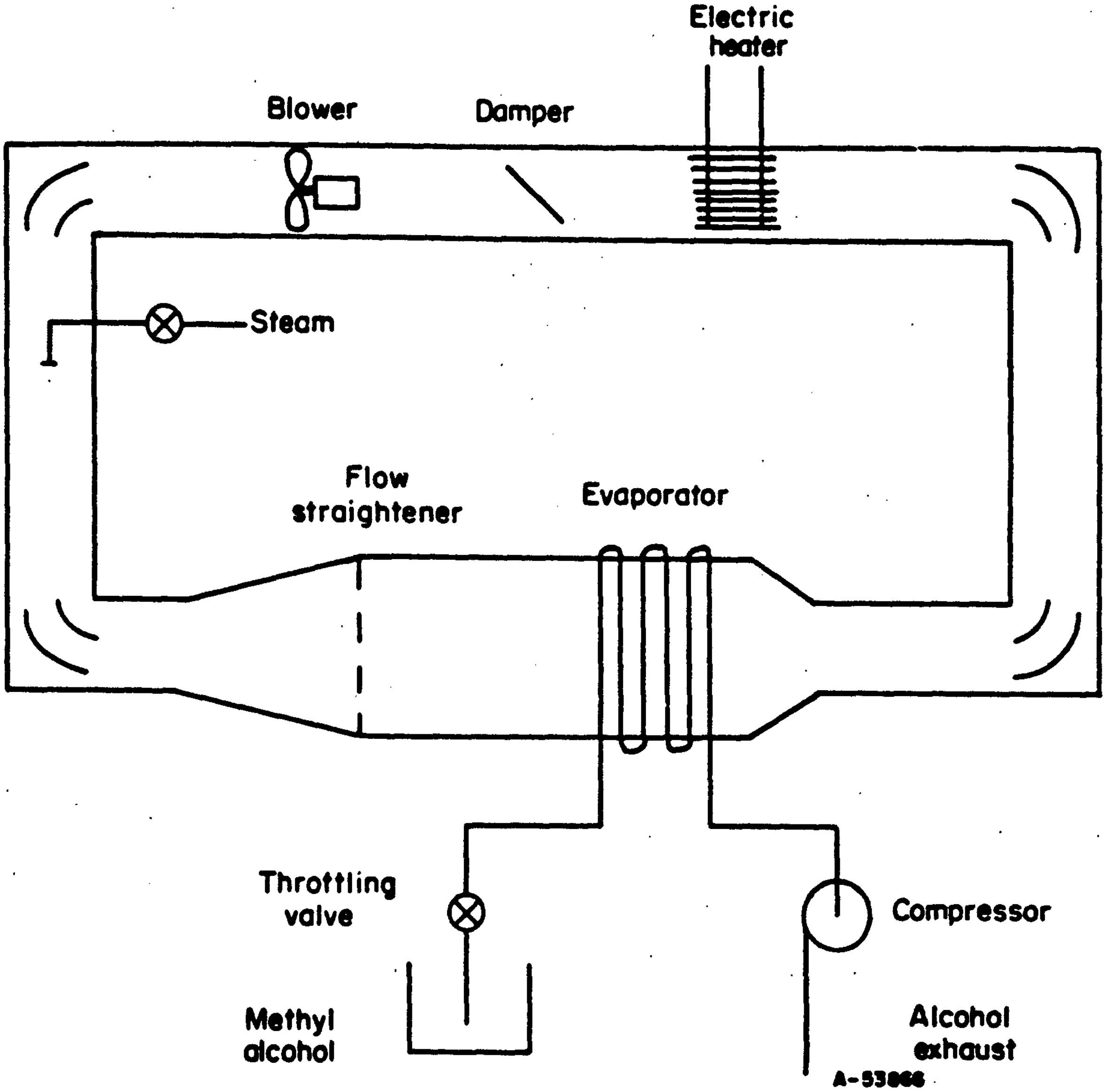

FIGURE 1. DIAGRAM OF LABORATORY COOLING SYSTEM

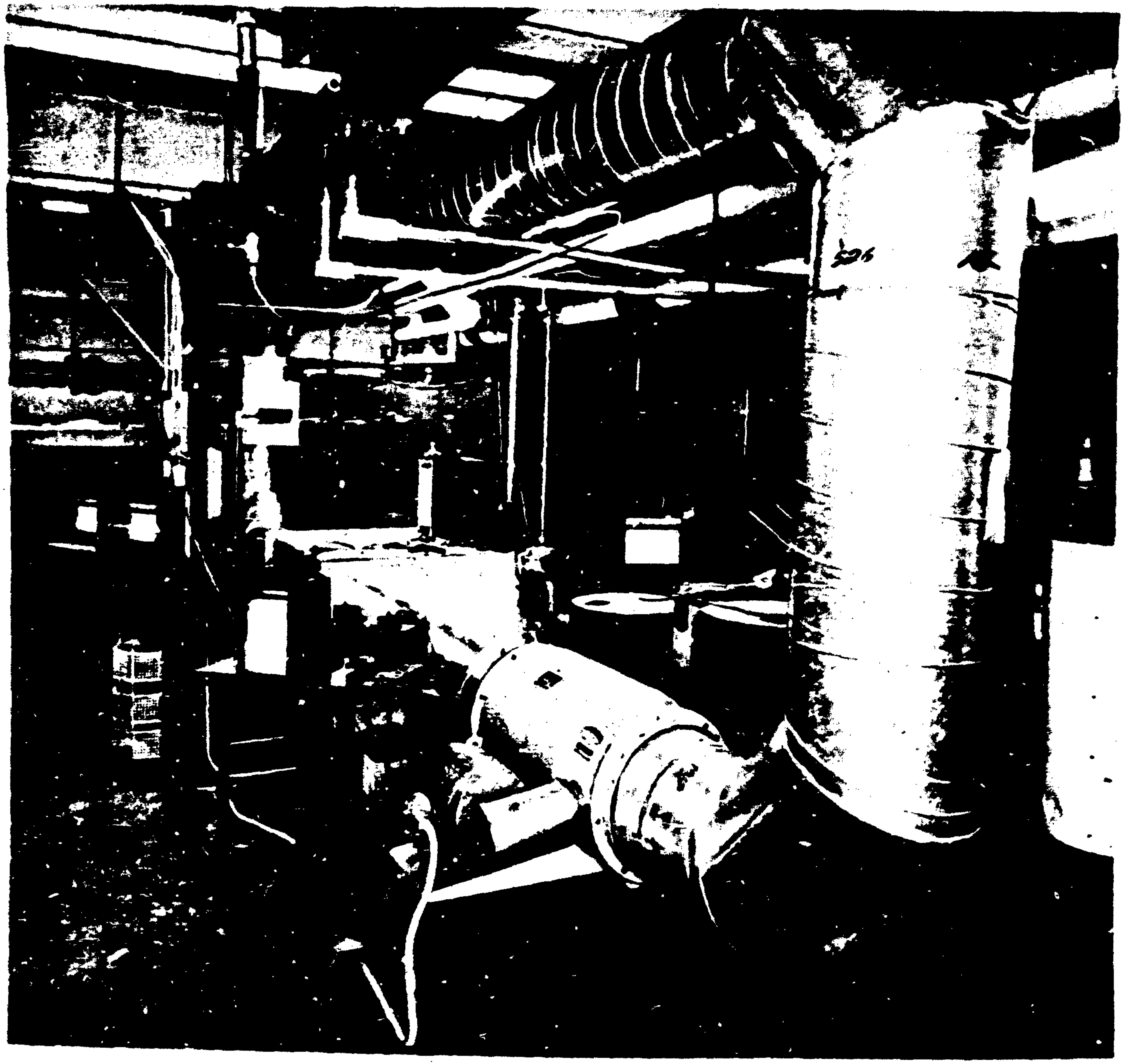

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FIGURE 2. LABORATORY COOLING SYSTEM

Evaporator

The evaporator was a commercial direct expansion unit of the type used for the direct expansion of refrigerants R-12 and R-22. Features of the coil were as follows:

30 by 36-in. face dimensions
20 tube circuits
6 rows deep
5/8-in.-OD copper tubes
Mechanically bonded aluminum fins
90 fins per foot
495 sq ft approximate external area
55 sq ft approximate internal area
188,000 Btu per hr capacity, standard air-conditioning rating.

Compressor

A Leiman 100-J vacuum pump was selected as the compressor for the methanol vapor after consideration of the features of 14 pumps listed in Table 1. The Leiman compressor was less costly and lighter in weight than any other of equal capacity. Also, the Leiman unit could be cooled by means of its lubricating oil and an air-cooled radiator, whereas the other choices would have required water cooling.

TABLE 1. VACUUM PUMPS CONSIDERED FOR COMPRESSION OF THE METHANOL VAPOR

Pumps	Speed, rpm	Rated Capacity, cfm, at 2.5 Psia	Rated Capacity, cfm, at 1.5 Psia	Power, hp	Type of Cooling	Weight, lb	Cost, $
Leiman							
100-J	900	112	110	7-1/2	Oil radiator	760	970
Vane							
Kinney							
KDH-130	535	118	118	5	Water	840	1760
KDH-250	475	222	222	10	Water	2100	3730
Rotary piston							
Fuller							
V-15	1160	55	40	5	Water	460	1310
V-30	1160	127	107	10	Water	780	1930
V-60	1160	244	212	15	Water	1500	2570
Vane							
Beech-Russ							
SS 7W	275	95	92	7-1/2	Water	2150	1440
SS-9W	260	193	188	15	Water	3150	2220
Vane							
Stokes							
212-H	475	140	140	5	Water	950	1760
412	490	270	270	10	Water	1750	3210
Rotary piston							
Nash							
CL203	1750	--	110	10	Water	--	1250
CL403	1170	--	260	20	Water	--	2200

Pump body
Material - cast iron

Vanes
Number - 4
Material - phenolic resin

Bearings
Antifriction

Shaft seal
Mechanical

Cooling
By lubricant

Rated discharge temperature
140 F

Lubricant circulation
Pump section separator, and gravity
1-1/2 gal per min

Maximum vacuum
29.8 in. Hg

Horsepower requirement at 27 in. Hg vacuum
7.4 at 125 cfm and 1000 rpm

Specifications on the Leiman 100-J pump for speeds between 500 and 1400 rpm showed that the volume displacement was directly proportional to speed and that at 27 in. Hg vacuum, the ratio of pumping capacity to horsepower was about 17 cfm per horsepower. This pump is not recommended for speeds above 1000 rpm for industrial applications requiring long life. However, for a period as short as 2 weeks, the pump should perform satisfactorily at 1400 rpm.

Lubricant

Keystone's 2K5 x Lt lubricant was selected according to the compressor manufacturer's suggestion and after consultation with lubricant manufacturers. This lubricant is a petroleum-base fluid of high molecular weight as provided by a mixture of long-chain hydrocarbons. The grade corresponded to an SAE 30 oil.

Blower

The blower for the laboratory system was a No. 152 Centaxial fan rated at 4400 cfm at 4 in. of water pressure, 3450 rpm, and 7.9 brake horsepower. The fan was capable of providing an air velocity of 600 fpm at the face of the evaporator against the pressure drops of the laboratory test facility. This fan is not of the type which would be

than that needed to operate the test facility.

Power

Electric motors were used to drive the blower and the compressor. In a shelter installation where circumstances would require the use of this methanol cooling system, internal combustion engines would probably be used to drive the mechanical equipment.

Electric Heater and Steam

An electric heater and steam were used to provide the desired temperature and humidity at the upstream face of the evaporator.

Instrumentation

In general, the variables measured were temperature, pressure, flow, and electric power. Sixty thermocouples were installed and read out on potentiometric recorders. The thermocouple installation is briefly as follows:

14 on the pump and associated equipment
1 in the methanol at pump suction
1 in the methanol at the top evaporator tube inlet
6 on the bends of the top evaporator tube
7 on the bends of the tenth evaporator tube
6 on the top of the top row of evaporator tubes along the evaporator centerline
2 in the airstream on each side of the evaporator
1 in the airstream on each side of the heater
8 between the coils of the top row of evaporator tubes in the air stream
1 on the bottom of the evaporator inlet manifold
1 in the ambient air.

Wet and dry bulb thermometers were located on either side of the evaporator transition pieces, and on the discharge side of the fan. Dry bulb thermometers were located adjacent to the two evaporator faces.

Air velocity in the loop was measured with a pitot tube and an inclined water manometer. The methanol flow was rate set using a rotameter and was measured with a scale and stop watch.

The vacuum of the methanol system was measured at the pump, evaporator inlet, and evaporator outlet with mercury manometers.

The power delivered to the vacuum pump motor was measured with a three-phase recording wattmeter.

EXPERIMENTAL RESULTS

The laboratory test loop was arranged in such a way as to allow variation of air temperature, humidity, air flow, methanol pressure, and methanol flow. Prior to the experimental program, data were taken and equipment modified to assure nearly uniform air velocity at the evaporator face and equal distribution of methanol to the evaporator, as indicated by uniform temperatures at the numerous thermocouples. Tests were run with the intent of studying the effect of each variable separately while holding the others constant. The range of variation is restricted by the limitations of the physical system itself, i. e., a certain degree of system matching is required to obtain significant and useful data. The data obtained, however, was quite sufficient to allow an evaluation of the concepts under study.

A total of 34 tests were made and the selected data are presented in Table 2 to support the discussion of the results. The cooling load was calculated as the product of the methanol flow and its latent heat of vaporization; consequently the accuracy of the cooling capacity is unknown in those cases where the evaporator was completely flooded. The number of rows wetted by the methanol was determined by the temperature of the return bends of the tubes. The methanol temperature was measured by a thermocouple placed inside a tube near the outlet of the capillary tube.

Evaporator, Counterflow

The first 29 tests were completed using a counterflow evaporator and a compressor speed of 1000 rpm. The air temperature, humidity, and velocity and methanol temperature were varied for the performance determinations.

The relatively small difference in cooling capacity was obtained because the capacity of the compressor was the limiting factor. Therefore, the performance data of the evaporator are most meaningful when analyzed on the basis of minor variation in cooling capacity.

Tests 5, 6, and 7 show that the air temperature drop through the evaporator was about the same for various air temperatures at the inlet. This was accomplished by the use of more of the evaporator surface as indicated by more rows being wetted by the methanol.

Comparison of Tests 11 and 14 with Tests 5 and 6 shows that the cooling capacity was decreased only from about 25,000 Btu per hr to 17,000 Btu per hr with an air flow change from 4750 cfm to 1050 cfm. The significance of this is that the major barrier to heat flow was on the methanol film on the inside of the tubes.

Tests 12, 13, and 23 show how fewer rows of tubes were required to evaporate the methanol as the methanol pressure and temperature were decreased. Qualitatively this is because of the greater temperature difference for heat transfer and higher heat-transfer coefficients on the methanol side due to the higher methanol velocities at the lower pressures. For these tests, the average overall heat-transfer coefficient was 3.9 Btu per hr per F log mean temperature difference per sq ft of evaporator surface.

TABLE 2. SELECTED TEST DATA

Test	Air Temperature, F — Inlet d.b.	Inlet w.b.	Inlet E.T.	Outlet d.b.	Outlet w.b.	Outlet E.T.	Air Flow, cfm	Methanol Press, psia	Methanol Temp, F	Methanol Flow, lb/min	Cooling Capacity, 1000 Btu/hr	Temp Pump Discharge, F	Pump Brake Horsepower, hp	No. of Rows Wetted by Methanol
Counterflow Evaporator, Compressor Speed, 1000 rpm														
5	90.0	86.0	86	85.0	84.3	84	4750	1.69	66.4	0.820	25.0	179	--	2-3
6	87.2	84.0	85	83.5	82.2	83	4750	1.62	66.4	0.820	25.0	180	--	3-4
7	84.9	81.1	82	79.8	79.0	79	3530	1.52	64.2	0.820	25.0	178	--	4-5
11	90.8	82.0	85	78.2	78.2	78	1050	1.61	66.4	0.571	17.4	218	--	3-4
12	90.0	82.0	85	80.0	79.5	80	1050	2.03	72.2	0.518	15.7[a]	218	--	6
13	89.1	81.7	85	78.7	79.3	79	1050	1.16	52.8	0.457	14.0	218	--	1-2
14	96.5	77.2	85	74.0	71.0	73	1050	1.67	64.7	0.574	17.5	--	6.15	4-5
21	93.0	81.5	85	77.0	75.4	75	800	1.62	63.7	0.562	17.1	226	5.95	4-5
23	91.0	80.0	87	78.0	76.0	81	1220	1.52	62.0	0.562	17.1	219	5.95	2-3
Concurrent Flow Evaporator, One-Half Evaporator Used, Compressor Speed, 1450 rpm														
31	92.7	80.2	85	74.0	73.5	74	800	1.59	63.6	0.697	21.2	256	11.2	4-5
32	93.0	81.9	86	77.0	77.0	77	800	1.41	59.7	0.601	18.4	249	11.0	2-3
33	92.2	81.0	85	76.0	75.5	76	800	1.44	66.0	0.647	19.7	248	11.0	3-4
34	93.0	81.2	85	76.5	76.0	76	800	2.05	73.6	0.982	29.8[a]	224	11.0	6

Note: Pressure drop on air side approximately 0.9 in. of water at 4750 cfm.

(a) Accuracy unknown when evaporator is flooded.

Evaporator, Concurrent Flow

In order to investigate the possibilities of further increasing the methanol film coefficients, three changes were made to increase the methanol velocity. The evaporator was so installed in the test loop that the air entered over the tubes where the methanol was admitted, and the methanol was admitted to only the top 10 of the 20 tub The reason for these changes was to achieve maximum velocity and evaporation as ea in the methanol tubes as possible. The other change was to increase the compressor capacity by running it at 1450 rpm rather than 1000 rpm.

Tests 31 and 33 showed an average cooling capacity of 20,500 Btu per hr at a methanol temperature of 62 F, and a total air flow of only 800 cfm. If four rows of th entire evaporator could have been used, the cooling capacity would have been 41,000 I per hr. This is 2.4 times the 17,000 Btu per hr of Test 21 which was run at an equal air flow rate and at about the same methanol temperature. Thus the concurrent-flow evaporator heat-transfer rates would be at least 2 times those for the counterflow arrangement, that is 7.8 and 11.0 Btu per hr per log mean temperature difference per sq ft of total area at velocities at the face of 133 and 630 ft per min, respectively.

Figures 3 through 6 show the temperature profiles obtained in the last four tests The curve labeled elbow was obtained by drawing a straight line between the temperatures measured by the thermocouples attached to the evaporator return elbows. These temperatures were close to the methanol temperature when there was a significant amount of liquid methanol in the tube. When the methanol was mostly vapor, heat tran fer from the methanol to the tube was small and the elbow temperatures approached ambient air temperature. Thus the elbow curve can be used as an indication of where vaporization occurs in the evaporator tube; a sudden increase in elbow temperature occurred when vaporization was complete. Of the four tests, complete vaporization occurred earliest in Test 32, then in Test 33, next in Test 31, and possibly not at all in Test 34.

The data for the curve labeled air in Figures 3 through 6 was obtained from the thermocouples located between the tubes in the top row of the evaporator. These show the variation of air stream temperature with evaporator tube length.

The four figures demonstrate the system matching briefly mentioned earlier. In Test 31 the methanol vaporized shortly after entering the tube because of the large temperature difference for heat transfer and high heat-transfer coefficients resulting from high-vapor velocities at low pressure. When the methanol flow rate was increased to obtain a higher cooling rate with no increase in compressor capacity, as in Test 33, th methanol pressure increased. This reduced the temperature difference and velocity in the tubes; consequently more tube length was needed to evaporate the methanol. This trend continues into Test 31. In Test 34, however, the flow and pressure was too grea and heat transfer fell off since the temperature difference was so small. This gives an indication of the matching of evaporator and pump required to obtain the most efficient combination of flow, pressure, and temperature difference, i.e., optimum heat transfer.

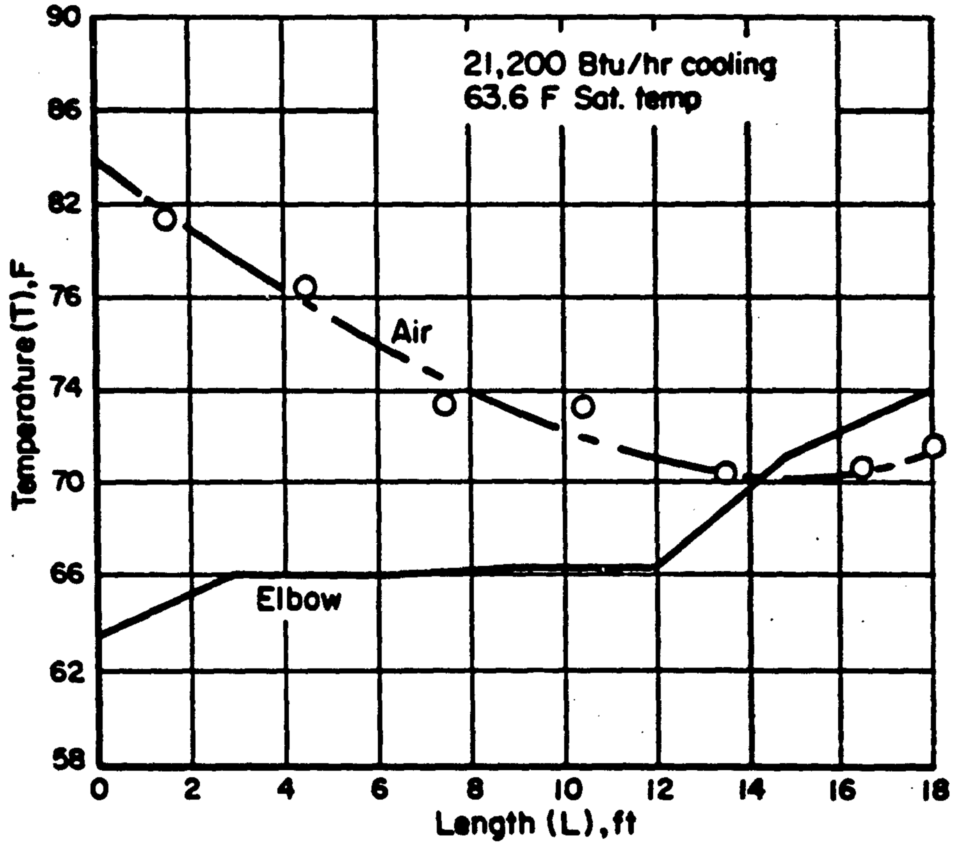

FIGURE 3. TEMPERATURE DISTRIBUTION – TEST 31

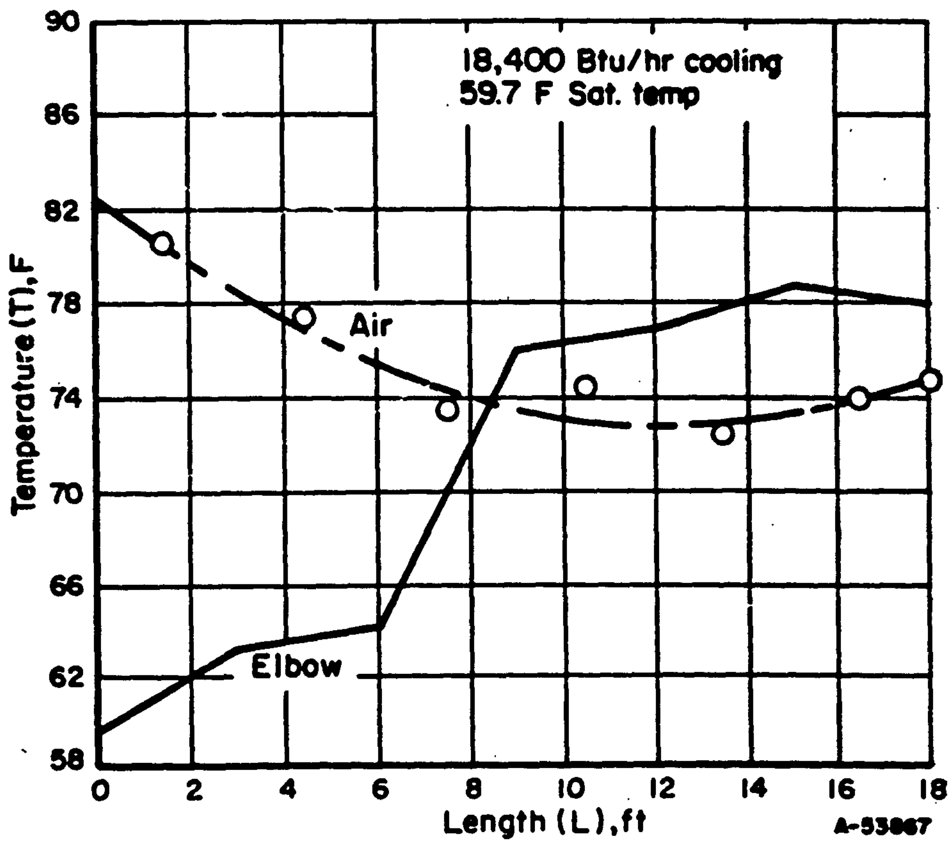

FIGURE 4. TEMPERATURE DISTRIBUTION – TEST 32

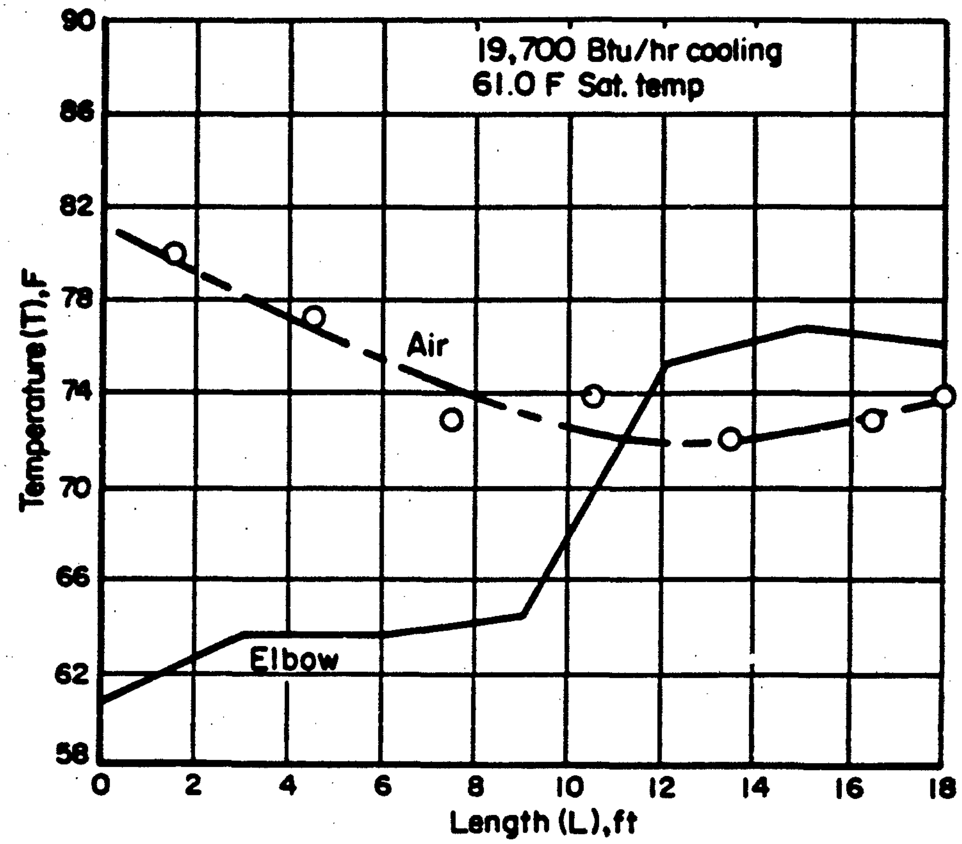

FIGURE 5. TEMPERATURE DISTRIBUTION – TEST 33

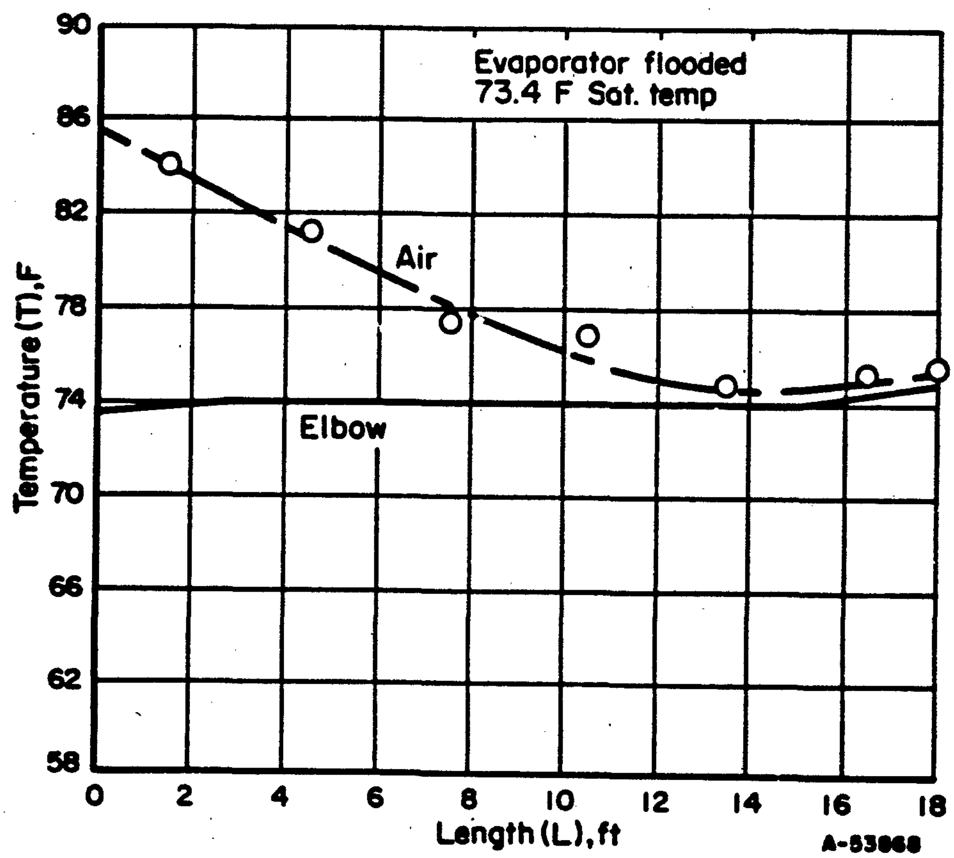

FIGURE 6. TEMPERATURE DISTRIBUTION – TEST 34

Compressor

Vapor pressure and temperature measurements at the compressor inlet in conjunction with the methanol flow rate verified that the compressor's volumetric capacity was equal to that specified for air by the manufacturer. The volumetric flow rate and equivalent cooling capacity when pumping methanol vapors are given in Table 3.

TABLE 3. VOLUMETRIC FLOW RATE AND EQUIVALENT COOLING CAPACITY OF 100-J LEIMAN COMPRESSOR

Pressure at Pump Inlet, psia	1000 Rpm Cfm	1000 Rpm Btu/Hr	1400 Rpm Cfm	1400 Rpm Btu/Hr
0.5	112	9,700	160	13,900
1.0	118	20,300	165	28,500
1.5	120	31,000	167	43,500
2.0	122	41,200	169	58,500
2.5	125	54,000	170	73,500

In a real installation, the piping connecting the compressor and the evaporator would be designed for a pressure drop of less than 0.1 psi; thus the pressure at the compressor inlet would be essentially the same as the evaporator pressure. For the laboratory work, the piping at the pump inlet included a valve for flow control and a filter which resulted in pressure loss and reduced the compressor's mass flow capacity.

Figure 7 shows how the brake horsepower requirement changed with pump temperature. Using the lubricant which was the equivalent to an SAE 30, the power reduced substantially as the pump temperature increased to 215 F because of reduced viscosity. Further increase in temperature resulted in higher horsepower because of binding in the compressor as the rotor expanded more than the cylinder. For shelter applications, the high initial starting power could require an oversized motor generator set if electric motors were used; on the other hand, if an internal-combustion engine was the prime mover, the compressor could be driven at rated power during the warm-up period by the reduction of speed. The power required at 140 F matched the rate published in the manufacturer's catalogue. As discussed later, the heat equivalent of the power input would have to be rejected to an appropriate heat sink.

At a gas discharge temperature of 180 F compressor housing temperatures were from 110 F at the inlet to 177 F near the outlet. None of the nine thermocouples on the pump indicated any hot spots.

In a cold startup, some of the methanol vapors would condense in the compressor, be separated with the recirculated lubricant from the remaining vapor, and drained to the lubricant sump. After the compressor and lubricant temperature increased above the boiling point of the methanol, the methanol would evaporate from the sump and the compressor would run free of all liquid except the lubricant.

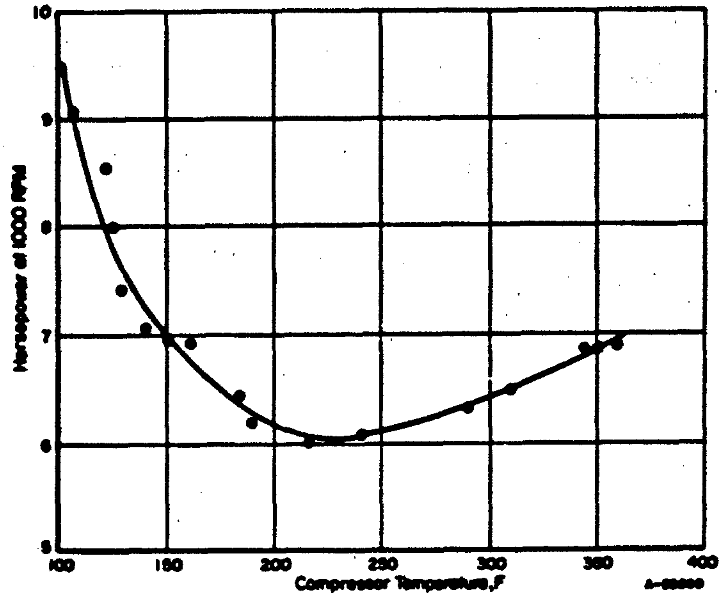

FIGURE 7. BRAKE HORSEPOWER REQUIREMENT AT 1000 rpm AND VARIOUS COMPRESSOR TEMPERATURES

Lubricant

The Keystone 2K5 x Lt lubricant was completely suitable for operation of the vane compressor when pumping methanol vapors. As discussed earlier on a cold startup, liquid methanol collected in the lubricant sump, but there was a clear line of separation between the lubricant and the methanol.

The lubricant used for the laboratory work discolored slightly but the adequacy of the lubricant was verified by no significant change in the viscosity.

OPTIMUM EVAPORATOR PRESSURE AND SYSTEM COSTS

The optimum evaporator pressure is that which would allow the lowest cost system. The components which have costs dependent upon evaporator pressure are the evaporator, compressor, blower, and power. To study these costs, estimates were based on the cost of equipment purchased for this laboratory work, flow rates, temperatures, and heat-transfer coefficients from the laboratory tests, and where necessary the costs shown in the first Battelle report on shelter cooling. The assumptions and costs used to evaluate the relation between the system cost and the evaporator pressure and temperature were:

1. Evaporator, \$0.90 per sq ft of heat-transfer surface

2. Compressor, \$5.90 per cfm gas at inlet at 1400 rpm

3. Heat-transfer coefficient
   7.8 Btu/(hr)(sq ft)(F) diff. at 160 fpm face velocity
   11.4 Btu/(hr)(sq ft)(F) diff. at 630 fpm face velocity

4. Air entering evaporator
   85 F effective
   93 F dry bulb
   80 F wet bulb

5. Air leaving evaporator
   74 F effective
   74 F dry bulb
   73-1/2 F wet bulb

6. Alcohol, $30 per 1,000 Btu per hr for 2 weeks

7. Alcohol storage per 1,000 Btu per hr for 2 weeks
   $12, fallout shelter
   $54, 30-psi blast shelter
   $94, 100-psi blast shelter.

Figure 8 shows how costs vary for the components which are dependent upon the methanol temperature and pressure and air flow rate. The lowest cost of evaporator, compressor, blower, and engine is about $48 at a suction pressure of about 1.5 psia, saturation temperature of 62 F.

The total cost of the complete system must include the components of Figure 8 plus the methanol, methanol storage, and means for cooling the engine and compressor. At 1400 rpm, the compressor had a cooling capacity of 43,500 Btu per hr at a suction pressure of 1.5 psia and a power input of 11 horsepower, equivalent to 28,000 Btu per hr. This power plus an equivalent amount for engine jacket cooling for a total of 56,000 Btu per hr would have to be rejected to some heat sink. The acceptable heat sinks for cooling the compressor and engine are air or water. Where the situation would permit, air would probably be the most convenient and would cost the least. However, if the system had to be completely isolated from the atmosphere, water would be the only other choice. This methanol system is considered primarily for situations where adequate quantities of natural water would not be available and, therefore, the water from limited natural supplies or from provided storage would have to be utilized by evaporation. Fortunately, as was demonstrated during this study, the compressor can operate at temperatures above the boiling point of water. Thus both the compressor and engine could be cooled by boiling water and ejecting the steam to the atmosphere.

Table 6 shows the costs of complete methanol cooling systems. The costs for the air or natural water-cooled compressor and engine are nearly the same as those predicted during the first shelter-cooling study and, therefore, the methanol system still warrants consideration for cooling shelters when refrigeration would be necessary.

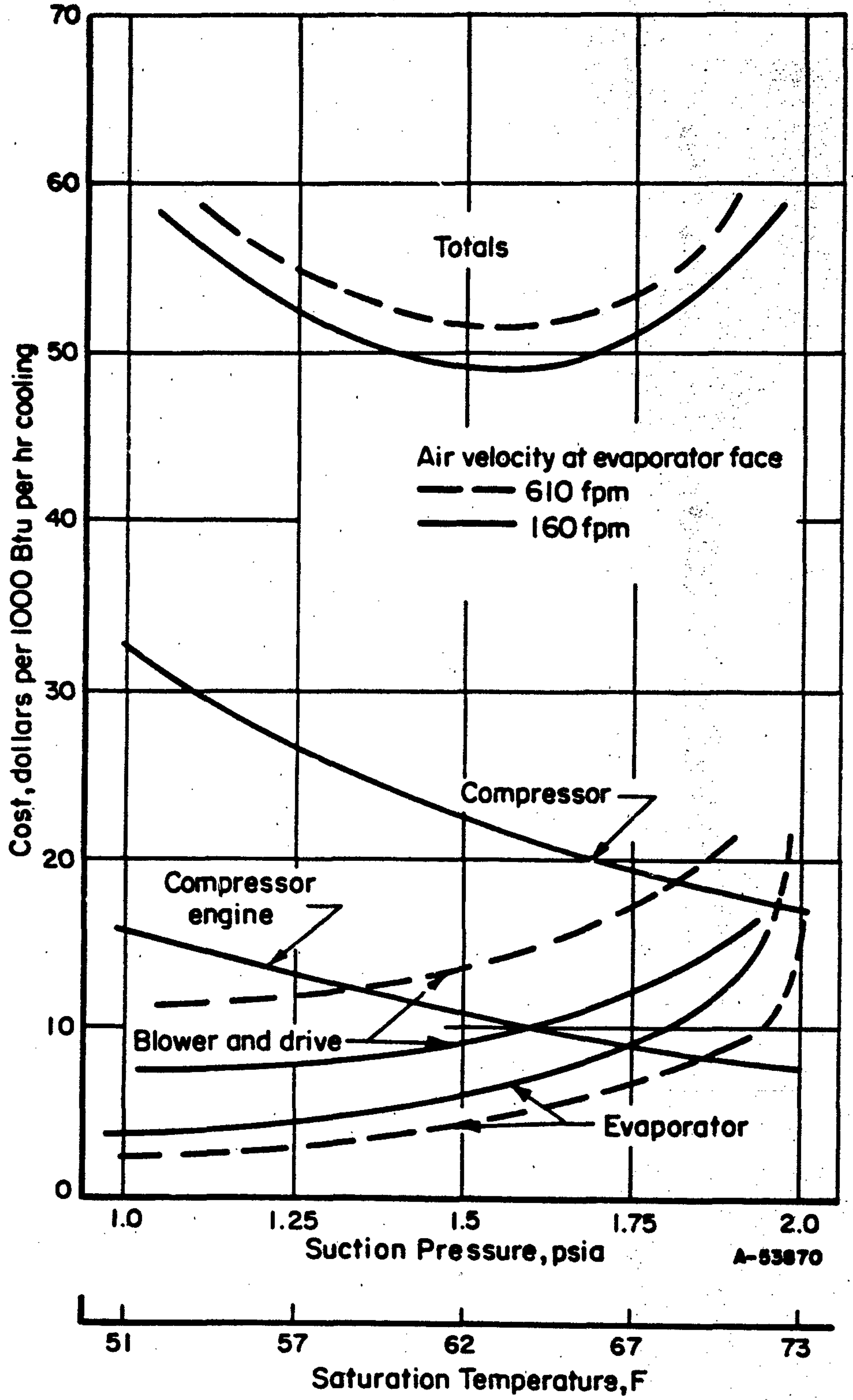

FIGURE 8. COSTS OF SYSTEM COMPONENTS AT VARIOUS SUCTION PRESSURES

TABLE 4. METHANOL COOLING SYSTEM COSTS

System Costs for 2-Week Operation,
Dollars per 1,000 Btu per Hr

	Fallout Shelter	30-Psi Blast Shelter	100-Psi Blast Shelter
Air or Natural Water-Cooled Compressor and Engine			
Compressor, Evaporator, Power, and Blower From Figure 8	48	48	48
Methanol	30	30	30
Methanol Storage	12	54	94
Total	90	132	172
Stored Water-Cooled Compressor and Engine			
Compressor, Evaporator, Power, and Blower From Figure 8	48	48	48
Methanol	30	30	30
Methanol Storage	12	54	94
Water Storage	8	35	61
Total	98	167	231

FUTURE WORK

The feasibility and reasonable cost of an open-cycle methanol cooling system has been demonstrated. The next step in the development program should be the assembly and operation of the complete system using the methanol vapors as fuel in an internal-combustion engine as the power source. The laboratory facility used for this study could be used as the refrigeration portion of the system to which would be added an engine with fuel metering for methanol and an appropriate means for cooling the engine and compressor.

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13. ABSTRACT

Methanol as the heat sink, for shelter cooling, was firmly demonstrated in a laboratory setup of a practical open-cycle cooling system. A commercially available rotary vane compressor, a lubricant nonmiscible with methanol, a direct-expansion evaporator, and a commercial methanol were employed to produce more than 25,000 Btu per hour of cooling at an effective temperature of 85 F.

Analysis of test data showed that the lowest system cost occurred at an evaporator pressure of 1.5 psia which corresponds to a saturation temperature of 62 F. Total system costs are comparable with those of alternative cooling systems.

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